DEVELOPMENT OF AN EQUIVALENT SOLID MODEL TO PREDICT THE VIBROACOUSTIC BEHAVIOUR OF EARMUFF CUSHIONS

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1. INTRODUCTION

Performances of earmuffs for hearing protection are largely affected by the seal between the earcup and the head [1]. The seal is done by a cushion, generally made of a foam piece in a polymeric sheath. In order to predict the sound transmission loss through earmuffs, it is necessary to model correctly both the mechanical and acoustical behaviour of the cushion. A lump model – which is generally used [2] – is sufficient at low frequency as it reproduces the whole body motion of the earcup with respect to the head. At higher frequency, this model is no longer appropriate as it does not account for sound transmission through the cushion material [3].

The aim of this work is to investigate the relevance of using an equivalent isotropic viscoelastic solid model to capture the sound transmission law of an annular cushion from a commercial earmuff.

In a first step, vibration analysis is done on a batch of ten cushions to (1) determine effects of a reproduced static force of the headband, (2) determine the roles of the holes drilled in the sheath, and (3) calculate the mechanical parameters (density, Young's modulus, Poisson's ratio and material damping) comparing vibration analysis results to a Finite Element model. Then these parameters are introduced in a combined Boundary Element – Finite Element (BEM-FEM) model to study the sound transmission loss through the lateral walls of the cushion.

2. VIBRATION ANALYSIS

Ten cushions from a hygienic spare set are studied. They are mounted on a shaker and submitted to the weight of three metal blocks with masses of 1.15kg, 1.65kg and 2.13 kg, representing the force applied by the head band when the earmuffs are positioned on the head. They are in the common range of tightening force [4]. A lighter fourth mass of 0.65kg is also used to test separately the foam and its protective sheath.

Transfer function H_1 between the acceleration of the mass and the acceleration delivered by the shaker is recorded for each experiment. As the first resonance peak is observed below 100 Hz, the frequency range of interest is from 10 to 100 Hz, with a frequency step of 125 mHz. The input signal chosen is a white noise, and 42 records are averaged per measurement. The transfer function is then compared to a 1-Degree of Freedom (1-DOF) system (mass-spring-damper). The comparison allows determining the frequency resonance (linked to the stiffness) and the material damping.

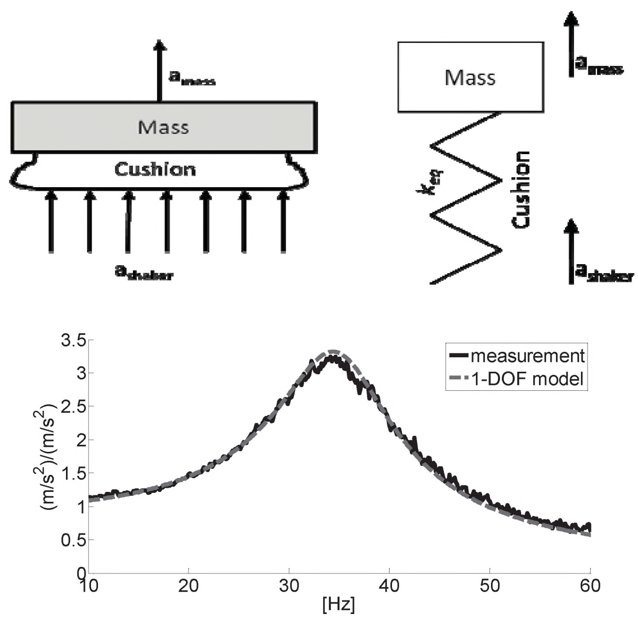

Figure 1. Vibration experiment result and its associated 1-DOF lump model.

2.1. Effect of mass and variation of the samples

For the three mass: 1.15 kg, 1.65 kg and 2.13 kg, the ten cushions are tested. Results presented Table 1 show that the frequency resonance decreases and the loss factor increases with the mass. The stiffness increases as the cushion is compressed. Equivalent stiffness of a 1-DOF system is obtained by the following formula: $k_{eq} = M\times 4\pi\times f_0^2$.

Table 1. Frequency resonances and loss factor, determined by fitting a 1-DOF system on data from measurements.

Mass M	1.15 kg	1.65 kg	2.13 kg
f_0 (Hz)	32 ± 0.9	31.8 ± 0.5	28 ± 0.5
η	0.17 ± 0.004	0.18 ± 0.004	0.20 ± 0.005
k_{eq} (N/m)	55,695	65,913	66,020

2.2. Effect of holes

The cushions have two perforations in the sheath. Effects of these holes are investigated in two ways:

1) Holes are glued with cyanoacrylate glue, when each cushion was under compression, to evacuate air from it. Vibration experiments were conducted and then cushions were immerged into water to make sure that apertures were completely occluded. Such verification has demonstrated that the sheath is a microporous material: micro air bubbles appear around the sheath when cushion are pressed. No bubble, hence no air, went out from the glued holes. Later, all glued cushion recovered their originally uncompressed form. It was then found that it is unnecessary to constrain the cushion before closing its holes, as after a time long enough, inner air will be evacuated by the microporous sheath. It was also found that the use of electrical tape provides a good occlusion of the holes, hence using glue is not necessary.

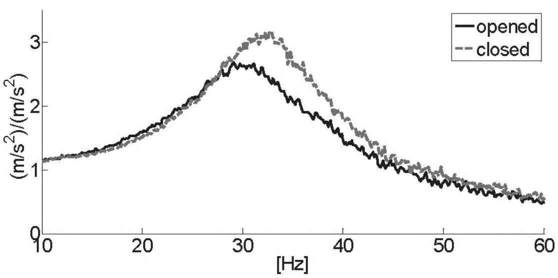

Figure 2. Effect of occluding holes. Experiment performed with a mass of 1.65kg.

Results presented Figure 2, for a mass of 1.65kg show that closing holes adds stiffness and decreases the loss factor, by introducing an air cavity. Similar results are obtained with a 1.15kg mass. Experiments were performed on three different cushions giving the same behaviour.

2) Holes are added around the cushion. A total of 14 holes of 1mm diameter were drilled progressively. Results can be observed on Figure 3. Resonance frequencies and amplitudes (as well as phases) of the transfer function are not changed when more than two holes are present. Consequently adding holes beyond the standard ones do not change significantly the dynamic behaviour of the cushion.

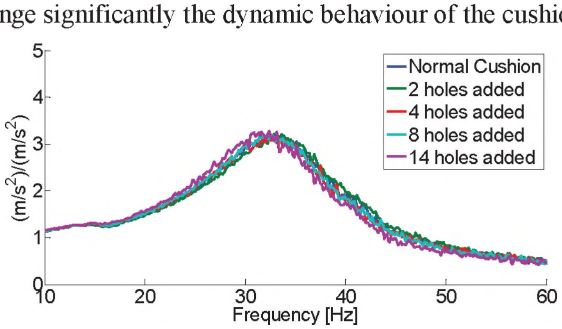

Figure 3. Effect of adding holes. Experiment performed with a mass of 1.65kg.

2.3. Foam and sheath stiffness separation

The foam and the sheath are tested separately under a mass of 0.65kg and compared to a cushion. This mass is chosen to avoid hysteretic compression of the foam. The sheath is obtained by cutting through the side of the cushion that is glued to the earcup. Loss factor and equivalent stiffness k_{eq} calculated as previously are presented in Table 2. The sheath has a dominant stiffness over the foam but its value is surprisingly greater that the combination of the two, this is attributed to the fact that the foam gives an initial curved shape to the sheath, hence reducing its buckling strength.

Table 2. Mechanical parameters of a cushion and its component, for a mass of 0.65kg

	Cushion	Sheath	Foam
f_0 (Hz)	36.2	44.8	14.8
η	0.165	0.214	0.105
k_{eq} (N/m)	33,627	51,503	5,621

3. OUTLINE OF THE FOLLOWING STEPS OF THE DEVELOPMENT OF EQUIVALENT SOLID MODEL

Results from the vibration analysis will be used to determine the mechanical parameters of an equivalent solid modelled with a FEM method [3]. We expect to obtain a set of Young's modulus (E) and Poisson's ratio (ν) pairs. Each (E, ν) pair will be then introduced in a BEM-FEM acoustic model to determine sound transmission loss through the lateral walls of the cushion when it is compressed, compare it to experimental data and determine the optimal (E, ν) pair. The results will be given during the oral presentation.

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